PCT

WORLD INTELLECTUAL PROPERTY ORGANIZATION
International Bureau

INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 6 : F16F 7/108, 1/371, 7/116

A1

(11) International Publication Number: WO 97/11288

(43) International Publication Date: 27 March 1997 (27.03.97)

(21) International Application Number: PCT/GB96/02332

(22) International Filing Date: 18 September 1996 (18.09.96)

(30) Priority Data:
9519118.5 19 September 1995 (19.09.95) GB

(71) Applicant (*for all designated States except US*): UNIVERSITY OF SOUTHAMPTON [GB/GB]; Highfield, Southampton SO17 1BJ (GB).

(72) Inventor; and
(75) Inventor/Applicant (*for US only*): PINNINGTON, Roger [GB/GB]; 44 Pointout Road, Bassett, Southampton SO16 7DN (GB).

(74) Agents: AYERS, Martyn, Lewis, Stanley et al.; J.A. Kemp & Co., Gray's Inn, 14 South Square, London WC1R 5LX (GB).

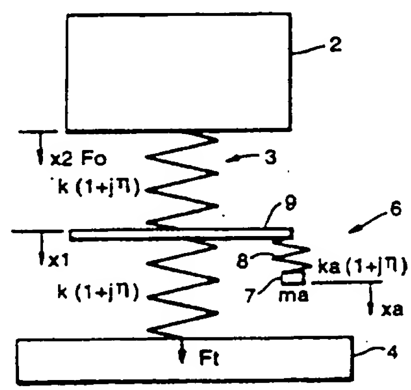
(81) Designated States: AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, CA, CH, CN, CU, CZ, DE, DK, EE, ES, FI, GB, GE, HU, IL, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MD, MG, MK, MN, MW, MX, NO, NZ, PL, PT, RO, RU, SD, SE, SG, SI, SK, TJ, TM, TR, TT, UA, UG, US, UZ, VN, ARIPO patent (KE, LS, MW, SD, SZ, UG), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, ML, MR, NE, SN, TD, TG).

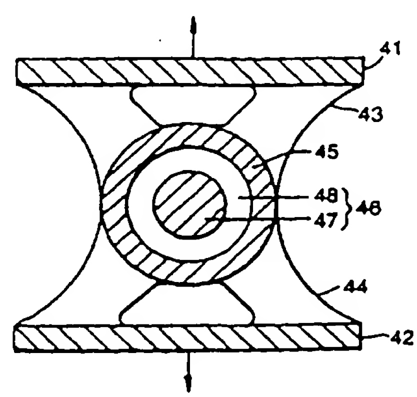
Published
With international search report.
Before the expiration of the time limit for amending the claims and to be republished in the event of the receipt of amendments.

(54) Title: VIBRATION ISOLATION DEVICE WITH A DYNAMIC DAMPER

(57) Abstract

A vibration isolation device comprises an isolator (3) for coupling the object (2) and the base (4) together, and a neutraliser (6) coupled into the isolator. A dip in transfer dynamic stiffness occurring at the natural frequency of the neutraliser (6) may be tuned to a characteristic frequency of the object (2) or to a natural frequency of the isolator-object (3, 6) system, in particular where the isolator (3) is a two stage isolator. The neutraliser (46) may be a spherical mass (47) embedded in rubber (48) within a hollow isolator mass (45) itself connected between two isolator elements (43, 44).





FOR THE PURPOSES OF INFORMATION ONLY

Codes used to identify States party to the PCT on the front pages of pamphlets publishing international applications under the PCT.

AM	Armenia	GB	United Kingdom	MW	Malawi
AT	Austria	GE	Georgia	MX	Mexico
AU	Australia	GN	Guinea	NE	Niger
BB	Barbados	GR	Greece	NL	Netherlands
BE	Belgium	HU	Hungary	NO	Norway
BF	Burkina Faso	IE	Ireland	NZ	New Zealand
BG	Bulgaria	IT	Italy	PL	Poland
BJ	Benin	JP	Japan	PT	Portugal
BR	Brazil	KE	Kenya	RO	Romania
BY	Belarus	KG	Kyrgystan	RU	Russian Federation
CA	Canada	KP	Democratic People's Republic of Korea	SD	Sudan
CF	Central African Republic			SE	Sweden
CG	Congo	KR	Republic of Korea	SG	Singapore
CH	Switzerland	KZ	Kazakhstan	SI	Slovenia
CI	Côte d'Ivoire	LI	Liechtenstein	SK	Slovakia
CM	Cameroon	LK	Sri Lanka	SN	Senegal
CN	China	LR	Liberia	SZ	Swaziland
CS	Czechoslovakia	LT	Lithuania	TD	Chad
CZ	Czech Republic	LU	Luxembourg	TG	Togo
DE	Germany	LV	Latvia	TJ	Tajikistan
DK	Denmark	MC	Monaco	TT	Trinidad and Tobago
EE	Estonia	MD	Republic of Moldova	UA	Ukraine
ES	Spain	MG	Madagascar	UG	Uganda
FI	Finland	ML	Mali	US	United States of America
FR	France	MN	Mongolia	UZ	Uzbekistan
GA	Gabon	MR	Mauritania	VN	Viet Nam

VIBRATION ISOLATION DEVICE WITH A DYNAMIC DAMPER

This invention relates to a vibration isolation device or arrangement.

A problem arises with objects such as machinery or buildings where vibration is transmitted between the movable object and a fixed object (base). This vibration can cause damage to the movable object or disturbance to people or machinery in its vicinity. One conventional method of reducing the transmission of vibrational forces between such objects has been by positioning a relatively flexible isolator between them. Typically the isolator would be made of rubber or a metal spring. This solution does not always adequately reduce the transmission of vibrations.

Another approach has been to apply a neutraliser (sometimes referred to as a dynamic absorber) to the movable object which is designed to provide a force to oppose the motion of the object. This approach is often used with buildings. The applied force from the neutraliser is proportional to its mass and acceleration. However, this has a problem in that the ratio of the mass of the neutraliser to the mass of the object must be about 1:10 to be effective and this can result in fatigue failure in the neutraliser itself. Furthermore, the object, e.g. a building, must be designed to withstand this mass.

In accordance with one aspect of the present invention there is provided a vibration isolation device for reducing the transmission of vibrations between two objects, the device comprising an isolator providing a load path coupling the objects together, and a neutraliser comprising a neutraliser mass mounted to a flexible portion, wherein at least a part of the flexible portion of the neutraliser which is distal from the neutraliser mass is incorporated into the load path of the

isolator.

The device operates more efficiently than conventional devices because the force from the neutraliser is coupled into the load path of the isolator. Thus, the force required to drive the velocity at the coupling point to zero is much smaller when applied to the isolator, than when applied to the relatively massive machine.

Preferably, the isolator includes a flexible portion into which the flexible portion of the neutraliser is coupled.

The present invention proposes combining the features of both the isolator and the neutraliser such that the force transmitted between the object and the base is reduced. A dip in transfer dynamic stiffness at the natural frequency of the neutraliser occurs when the neutraliser is combined with the isolator. This dip is greatest when the mass of the neutraliser is large relative to the object, but satisfactory results are achieved when the mass of the neutraliser is less than 1/10th of the mass of the object.

The response of the device can be further improved by adjusting the damping of the isolator and/or neutraliser.

Conventionally, single stage isolators have been used. In some cases a two-stage isolator has been proposed which has a beneficial effect in that the response drops off at a higher rate with frequency above its resonance point than with a conventional single stage isolator so that high frequency vibrations are not easily transmitted between the object and the base. However, a disadvantage of the two stage isolator is that a resonance peak exists at the natural frequency of the two-stage system and so they are not used very often. The present invention makes the use of a two stage isolator possible because it enables the resonance peak to be reduced by combining a vibration neutraliser with a two-stage

isolator. By tuning the neutraliser natural frequency to the vicinity of the natural frequency of the two-stage system the resonance peak may be reduced.

Preferably, the neutraliser is configured to act in two perpendicular directions. This may be simply achieved by a symmetric configuration of the neutraliser. For example, the neutraliser mass may be disposed internally of a support in the isolator or of an isolator mass in a plural stage isolator.

Examples of types of neutraliser which could be used in the present invention are a double cantilever beam, a bolt embedded in an elastic material or a tensioned wire.

In accordance with another aspect of the present invention, there is provided a vibration isolation arrangement for reducing the transmission of vibrations between two interconnected objects, the arrangement having a mechanical equivalent circuit constituted by the masses of the two objects interconnected by the series coupling of a first and a second damped elastic compliance and a mass coupled via a third damped elastic compliance to the coupling point of said first and second damped elastic compliancies.

Examples of vibration compensation devices according to the present invention will now be described and contrasted with conventional use of neutralisers and isolators with reference to the accompanying drawings in which:

Figure 1	illustrates an example of a conventional vibration neutraliser;
Figure 2	illustrates the variation in force transmissibility with normalised frequency for a mass system having a simple isolator;
Figure 3	illustrates the variation in force transmissibility with normalised frequency for

the system and the neutraliser of Figure 1;

Figure 4 shows a measured variation of transfer dynamic stiffness with frequency of a conventional isolator;

Figure 5 illustrates a conventional two-stage isolator;

Figure 6 illustrates an example of a vibration compensation device in accordance with the present invention;

Figure 7 shows how the transfer dynamic stiffness of the device of Figure 6 varies with frequency;

Figure 8 illustrates a vibration compensation device in accordance with the present invention using a two-stage isolator;

Figure 9A&9B show how the transfer impedance and the dynamic stiffness vary with frequency for the devices of Figure 5 and Figure 8;

Figure 10 illustrates an example of the device of Figure 6 using a cantilever type neutraliser;

Figure 11 illustrates an example of the device of Figure 8 using a cantilever type neutraliser;

Figure 12 illustrates an example of the device of Figure 6 using a steel bar and masses as neutraliser;

Figure 13 illustrates an example of the device of Figure 6 using tensioned wire and masses for the neutraliser; and

Figure 14 shows a two stage isolator wherein the neutraliser is disposed within the isolator mass.

In the following description reference will be made to force transmissibility which is the ratio of the force transmitted (Ft) to the object (or base) at a particular frequency to the force applied (Fo) by the base (or object); transfer dynamic stiffness which is the ratio of the force transmitted (Ft) to the machine displacement (X); and transfer impedance which is the ratio of force transmitted (Ft) to velocity (V_2).

Figure 1 schematically illustrates a conventional vibration neutraliser 1 of mass Ma attached to an object, namely machine 2 of mass M. A conventional vibration isolator 3, in this example in the form of a spring, couples the machine 2 to a foundation 4. If the example of Figure 1 had only an isolator, then a peak value of force transmissibility occurs at the natural frequency of the isolator as illustrated in Figure 2 which shows the variation in force transmissibility with frequency (normalised by the natural frequency of the isolator, wn) for different isolator dampings.

When a neutraliser is used in a conventional configuration, the system has two resonances, above and below the neutraliser natural frequency which are not present in a system using an isolator alone. Figure 3 shows graphically the variation in force transmissibility with frequency (normalised by the natural frequency of the neutraliser, wa) for different stiffness ratios (neutraliser stiffness/isolator stiffness). In this example, the isolator and neutraliser damping is kept constant at 0.05 and the neutraliser natural frequency, wa is tuned to the natural frequency of the machine-mass system wn. A minimum response is achieved at the neutraliser natural frequency. From Figure 3 it can be seen that increasing the neutraliser stiffness implies an increase in the neutraliser mass. The lower the stiffness ration ka/k and hence mass ratio Ma/M, the closer together the resonance peaks. Thus the neutraliser is less effective as its mass decreases relative to the machine mass. To neutralise at a particular frequency it is best to have as large a neutraliser as possible. The transmitted force will still be reduced to a minimum at the neutraliser natural frequency, however a relatively small neutraliser mass will have a relatively large displacement.

In this conventional neutraliser system the transfer dynamic stiffness has a constant value equal to the stiffness of the machine isolator. No dip is observed at the natural frequency of the neutraliser. An experimental example of this is

illustrated in Figure 4 which shows the variation in transfer dynamic stiffness with frequency.

Figure 5 schematically illustrates a conventional two-stage vibration isolator. The machine 2, of mass M is coupled to the foundation 4 by an isolator 3. In this example the two stage isolator comprises two flexible elements, springs 10,11, which are connected via a 2 stg mass 5, Mi. As described above, the two stage isolator is often not used because of the resonance peak which occurs at the natural frequency of the system. This peak is shown in Figure 9A by curve 13 and in Figure 9B by the solid curve.

Figure 6 is a schematic diagram of an example of a vibration isolation device in accordance with the present invention. One object, namely machine 2 is coupled to another object, namely foundation (or base) 4 via an isolator 3 (which is one stage). A vibration neutraliser 6 consisting of a mass 7, Ma coupled via a spring 8 to a support 9 is positioned in the isolator 3. Thus the neutraliser 6 is coupled into the isolator 3 approximately mid-way between its ends, but it could be coupled closer to one end or the other.

Figure 7 shows how, for the device of Figure 6, the transfer dynamic stiffness of an example of a vibration compensation device varies with normalised frequency for different stiffness ratios. The isolator and neutraliser damping are constant at 0.3 and 0.05 respectively, while the stiffness ratio is varied. Compared to the response with no neutraliser shown by the horizontal line, a dip can be seen at the natural frequency of the neutraliser. This is preceded by a maximum. This phenomenom is independent of the natural frequency of the main object-isolator system. As the stiffness ratio decreases the preceding peak approaches closer to the dip and so the useful attenuation bandwidth is less. Hence a high ratio is desirable. As with a conventional neutraliser, it is best to have as large a neutraliser as possible but the system will

operate effectively with much smaller neutralisers than the conventional systems.

When a neutraliser 1 is mounted on the isolated machine mass system 2 in a conventional way, as in Figure 1, no such dip occurs in the dynamic stiffness/frequency response (Figure 4). A constant value equal to the stiffness of the isolator 3 is maintained. However, in the device of the present invention a neutraliser 6 is introduced at the isolator 3, with a natural frequency tuned to a characteristic frequency of the machine, for example to the running speed of the machine, resulting in the force transmitted to the foundations 4 being reduced. This is regardless of whether the machine 2 is resonating or not. To obtain a good isolation at low frequency it is necessary to have a soft isolator, sometimes leading to instability problems, however the same performance could be achieved at a single frequency with a stiff isolator combined with the neutraliser.

Figure 8 shows a vibration compensation device in accordance with the present invention and incorporating a two stage isolator. The mass 2 is coupled to the foundation 4 via a vibration isolator 3. The flexible portion of isolator 3 consists of two separate flexible elements, springs 15,16, each coupled to an intermediate mass 14 (sometimes called a blocking mass). The two stage mass 14 is mounted on the neutraliser support 9. The neutraliser mass 7 is coupled via spring 8 to the support 9. The flexible portion could include a series connection of further flexible elements separated by masses to make it plural-stage.

In Figure 9A, curve 12 shows the variation of transfer impedance with frequency of the device of Figure 8. The isolator and neutraliser damping is constant at 0.15 and 0.05 respectively, the stiffness ratio is 0.28 and the neutraliser natural frequency is tuned to the two stage frequency of the isolator (i.e. Wn=Wa). The use of a two stage isolator and

neutraliser in the device of the present invention produces a dip at the natural frequency and improved transfer impedance at other frequencies. The natural frequency of the neutraliser is tuned to coincide with the natural frequency of the two stage system resulting in a reduced resonance peak at the expense of introducing two new peaks either side of the original. On average however the response is less than the original.

Figure 9B shows the effect of changing the neutraliser properties to obtain some optimization. The two dotted lines in Figure 9B show, for two different configurations of the device of Figure 8, the variation of dynamic transfer stiffness with frequency, as compared to the conventional two stage mount shown by a continuous line. In the configuration shown by the dotted line: the neutraliser loss factor, $\eta a=1.6$; the stiffness ratio $ka/2k=0.4$; and the natural frequency ratio $f_a/f_I=0.5$. In the configuration shown by the dot-dash line: $\eta a=0.8$; $ka/2k=0.2$; and $f_a/f_I=0.7$. In both configurations the isolator loss factor, $\eta=0.2$.

Again the responses are less than the original. The shape of the curve can be altered by tuning the neutraliser as compared to the isolator-object system. Where the neutraliser is a mass coupled to a flexible portion which is a rubber mix, the properties may be selected by altering constitution of the rubber mix and the size of the mass.

Specific examples of types of vibration compensation device will now be described. These were constructed with three different designs of neutraliser. Figure 10 shows a double cantilever beam 19 coupled into isolator 23 made of rubber and acting as a spring with masses 17, 18 connected to each end.

For experimental purposes, the system is driven by an electrodynamic shaker 36 including magnet 37 and coil 38 coupled to one end of the isolator. An accelerometer and a

force transducer 39 measure the displacement and force produced by the shaker 36 at that one end. A further force transducer 40 measures the transmitted force, at the other end.

The cantilever beam acts as the neutraliser, the stiffness and natural frequency of which can be estimated using simple theory:

$ka = (3EI)/l^3$, where E = Youngs Modulus of beam; I = second moment of area of beam; l = effective length of the beam; and M = effective mass (approximately added mass + 1/3 of mass of the one half of the beam).

Then : $fa = (1/2\pi) . (ka/m)^{½}$

The effective length of the beam 19 was taken as the distance from the centre line 20 of the isolator 23 to the centre line 21 of the mass 17. Errors in this calculation arise from the fact that the boundary conditions at the supported end of the beam are not ideally clamped and the effect of the mass of the beam 19 is not properly considered. The latter effect is assumed to be negligible as the mass of the cantilever beam 19 is small compared to the added masses 17, 18. Note the above formula is for a single cantilever. For the double cantilever beam used in the experiment, the total stiffness of the neutraliser can be considered as twice that of a single beam.

In the experiment E = 207 GN/m^2 and I = $26x10^{-12}m^4$

From this the predicted neutraliser natural frequency and stiffness can be determined for a given neutraliser mass and cantilever length.

The stiffness ratio was varied by changing the neutraliser mass and adjusting it along the length of the beam so that the natural frequency stayed the same. As a comparison, if

neutraliser 19 was positioned at the top of the isolator 23 it would act as a conventional neutraliser.

A variation on this configuration is to use two double cantilever beams (not shown) with different natural frequencies. Two modes are advantageous. The first mode is with the natural frequency of the second beam equal to twice that of the first. This could be useful to reduce the force transmitted from a machine at both 1 x and 2 x RPM of the machine. The second mode is with the natural frequency of the second tuned to coincide with the response peak of the first to improve the response achieved.

In the example of Figure 11 incorporating a two stage isolator, a mass 24 of steel was positioned between two pieces of rubber 25, 26 to create a two stage isolator. The cantilever 19 was positioned in the mass 24 to act as a neutraliser. The neutraliser is tuned so that its natural frequency coincides with the natural frequency of the two stage isolator.

For the example of Figure 12 a small diameter steel bolt 27 is embedded in a thick rubber plate 28 which is the isolator by drilling a hole through the rubber and press fitting the bolt 27 through the hole. The stiffness of the rubber acting on the bolt acts as the neutraliser stiffness. The neutraliser mass and so natural frequency can be varied by adding masses 29 to the bolt.

In the configuration of Figure 13 a wire 30 is connected to a plate 31 at the centre of the isolator 33. Masses 32, 34 are connected to the wire 30 which is tensioned on a rigid frame 35. At its natural frequency the wire-mass system acts as a vibration neutraliser. The natural frequency can be varied by changing the wire tension using a threaded screw (not shown).

The symmetry of the configurations of Figures 12 and 13 allows

the respective neutraliser to act on vibrations polarised both parallel and perpendicular to the respective isolator.

Figure 14 shows, in cross section, a two stage isolator which is particularly suitable as an engine mount. In this isolator, the neutraliser is disposed inside the isolator mass which provides a compact structure. To create a two stage isolator, a mass 45 is connected between two flexible elements 43,44 each being a pair of legs made of rubber and being connected to a respective flange 41,42. The flanges 41,42 are for mounting the isolator to respective objects. The isolator mass 45 is a cylinder with a cylindrical hollow, and disposed therewithin is the neutraliser 46 consisting of a neutraliser mass 47 embedded in an elastomeric material 48 such as rubber. The circular symmetry visible in the cross section allows the neutraliser 46 to act on vibrations polarised in directions parallel and perpendicular to the flanges 41, 42.

Alternatively, the isolator mass 45 and neutraliser mass 47 may be spherical, whereby the neutraliser acts on vibrations polarised in all directions parallel to the flanges 41, 42, in addition to vibrations polarised perpendicular to the flanges 41,42.

Possible uses of the present invention are for an isolator tuned to the natural frequency of a building which is vulnerable to earthquake, or to suppress the natural frequency of a two-stage mount, so that it can be used under an automobile engine. A stiff mount could be used to provide a good quality ride while a double mount could be used to attenuate high frequency vibrations. Other transport uses are in ships and trains. Another application is removal of troublesome excitation frequencies, for example, the spinning of electrical machines or spin dryers where a stiff isolator could be used with a neutraliser within it tuned to the excitation frequency.

CLAIMS

1. A vibration isolation device for reducing the transmission of vibrations between two objects, the device comprising

an isolator providing a load path coupling the objects together, and

a neutraliser comprising a neutraliser mass mounted to a flexible portion, wherein at least a part of the flexible portion of the neutraliser which is distal from the neutraliser mass is incorporated into the load path of the isolator.

2. A vibration isolation device according to claim 1, wherein the isolator includes a flexible portion into which the flexible portion of the neutraliser is coupled.

3. A vibration isolation device according to claim 2, wherein the flexible portion of the isolator has two ends for connection, respectively, to the two objects, and said part of the flexible portion of the neutraliser is coupled approximately mid-way between said two ends of the flexible portion of the isolator.

4. A vibration isolation device according to claim 2 or 3, wherein the flexible portion of the isolator is made of an elastomeric material.

5. A vibration isolation device according to any one of the preceding claims, wherein the isolator is a plural-stage isolator.

6. A vibration isolation device according to claim 5, wherein the plural-stage isolator includes an isolator mass coupled between two flexible elements.

7. A vibration isolation device according to claim 6,

wherein the isolator mass is hollow and the flexible portion of the neutraliser is disposed therewithin.

8. A vibration isolation device according to claim 7, wherein the isolator mass is a cylinder.

9. A vibration isolation device according to any one of the preceding claims, wherein the flexible portion of the neutraliser comprises elastomeric material within which the neutraliser mass is embedded.

10. A vibration isolation device according to any one of the preceding claims, wherein a natural frequency of the neutraliser is tuned to the vicinity of a characteristic frequency of one object.

11. A vibration isolation device according to any one of claims 1 to 10, wherein a natural frequency of the neutraliser is tuned to the vicinity of a natural frequency of the isolator-object system.

12. A vibration isolation device according to any one of the preceding claims, wherein the neutraliser is configured to act on vibrations polarised in at least two perpendicular directions.

13. A vibration isolation device according to any one of the preceding claims, wherein the neutraliser damping is one twentieth of the isolator damping or greater.

14. A vibration isolation device according to any one of the preceding claims, wherein the neutraliser mass is one tenth of the mass of the lighter object or greater.

15. A vibration isolation device according to any one of the preceding claims, wherein the neutraliser is one of the group consisting of a cantilever beam, a double cantilever beam, a

tensioned wire supporting a mass, and a bolt embedded in elastomeric material.

16. A vibration isolation device according to any one of the preceding claims, being an engine mount for the engine of a vehicle.

17. A vibration isolation arrangement for reducing the transmission of vibrations between two interconnected objects, the arrangement having a mechanical equivalent circuit constituted by the masses of the two objects interconnected by the series coupling of a first and a second damped elastic compliance and a mass coupled via a third damped elastic compliance to the coupling point of said first and second damped elastic compliancies.

Fig.1.

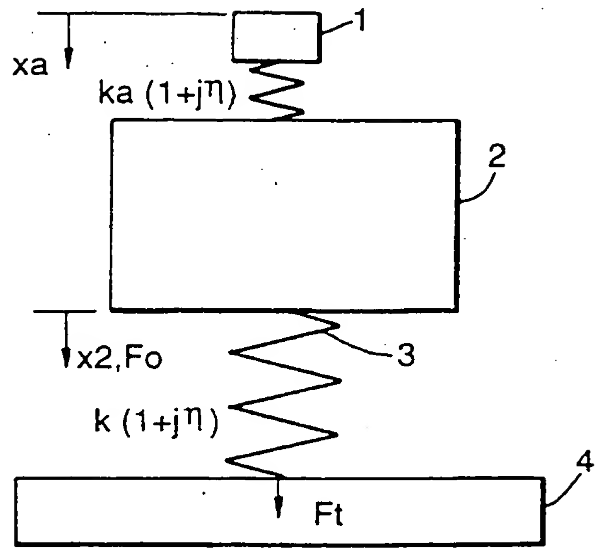


Fig.5.

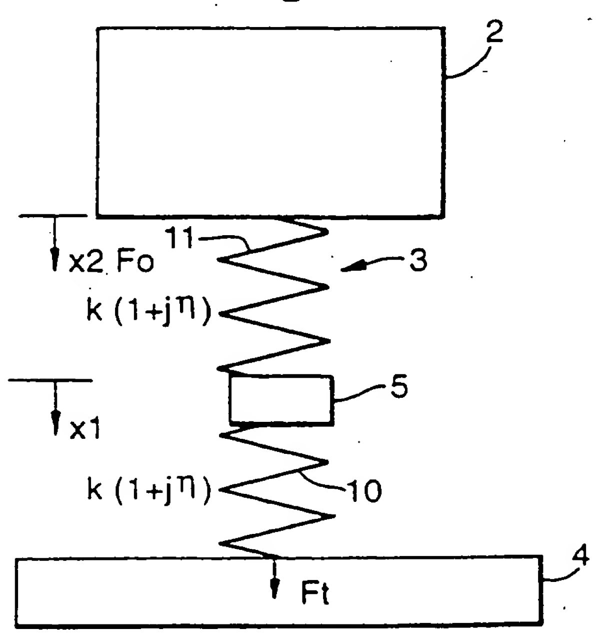


Fig.2.

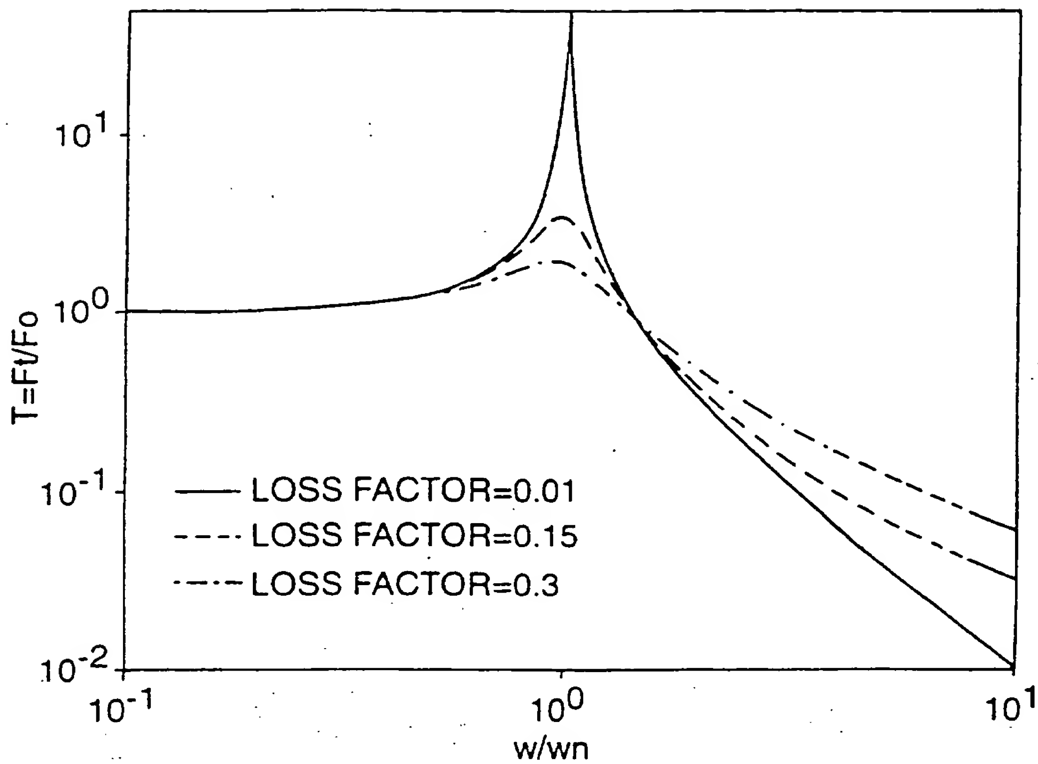

10^1
10^0
10^-1
10^-2
T=Ft/Fo
LOSS FACTOR=0.01
LOSS FACTOR=0.15
LOSS FACTOR=0.3
10^-1
10^0
10^1
w/wn

Fig.3.

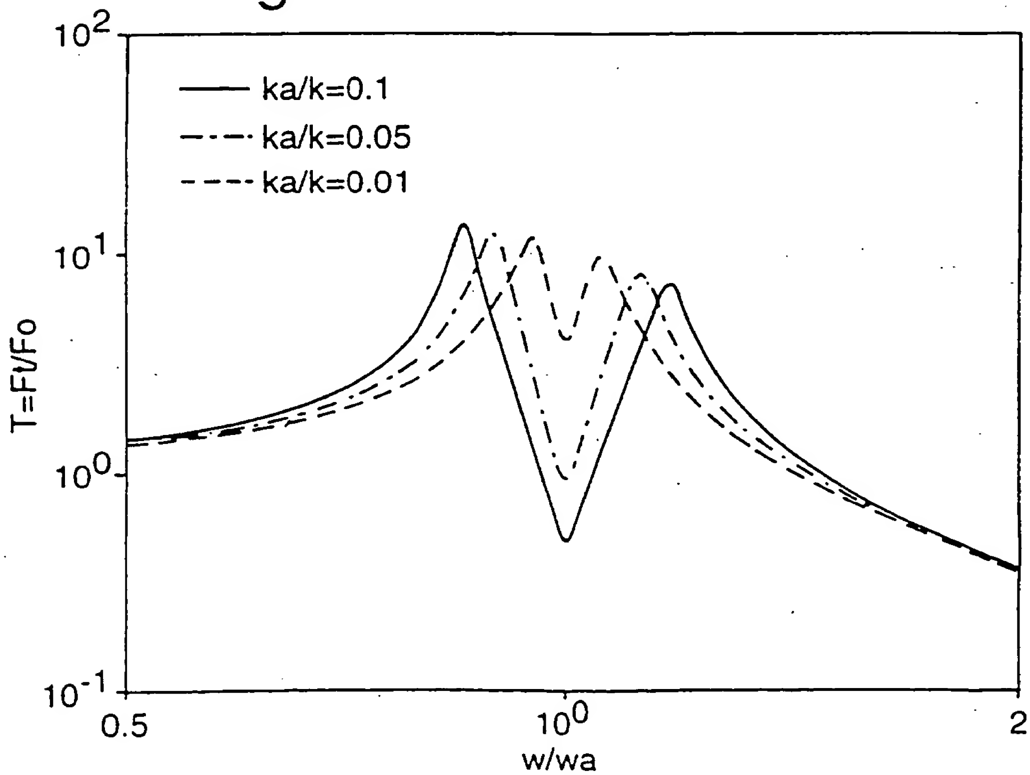

10^2
10^1
10^0
10^-1
T=Ft/Fo
ka/k=0.1
ka/k=0.05
ka/k=0.01
0.5
10^0
2
w/wa

Fig.4.

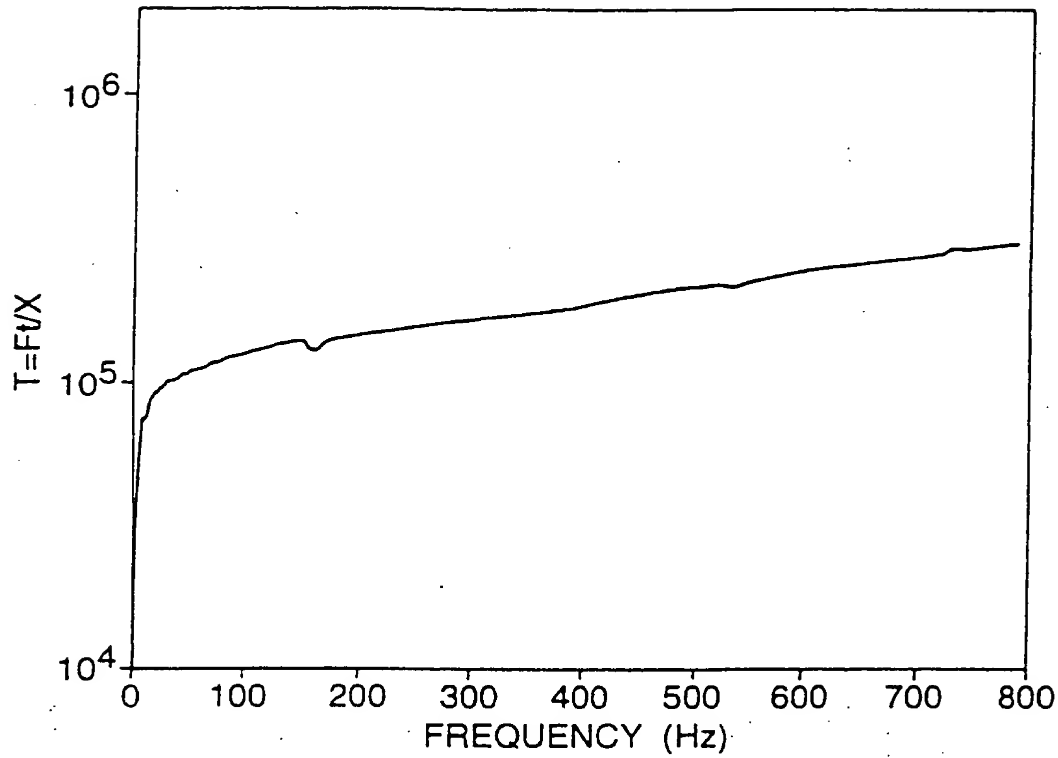

10^6
10^5
10^4
T=Ft/X
0
100
200
300
400
500
600
700
800
FREQUENCY (Hz)

Fig.7.

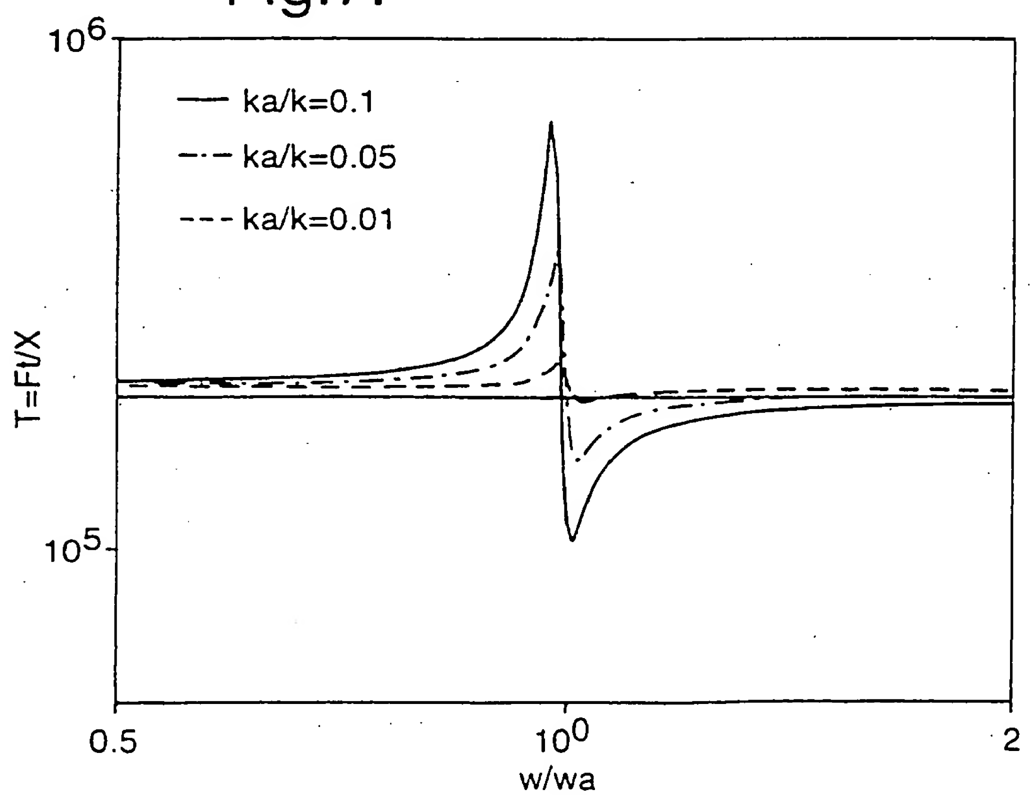

10^6
10^5
ka/k=0.1
ka/k=0.05
ka/k=0.01
T=Ft/X
0.5
10^0
2
w/wa

Fig.6.

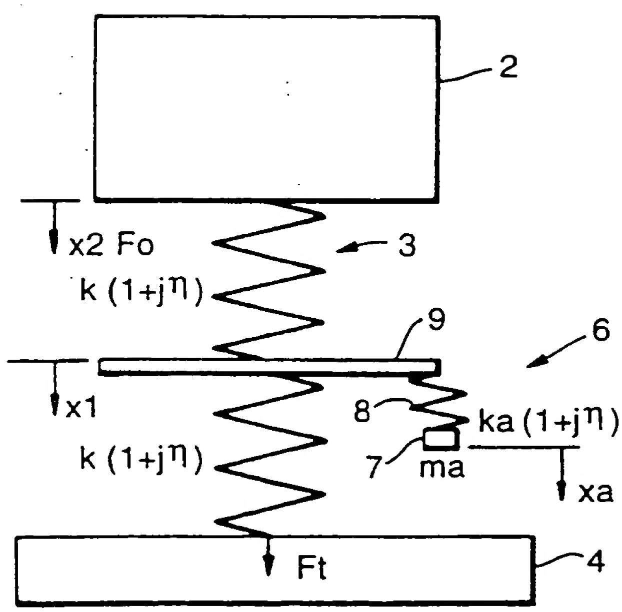

2
x2 Fo
3
k (1+jη)
9
6
x1
8
ka (1+jη)
k (1+jη)
7
ma
xa
Ft
4

Fig.8.

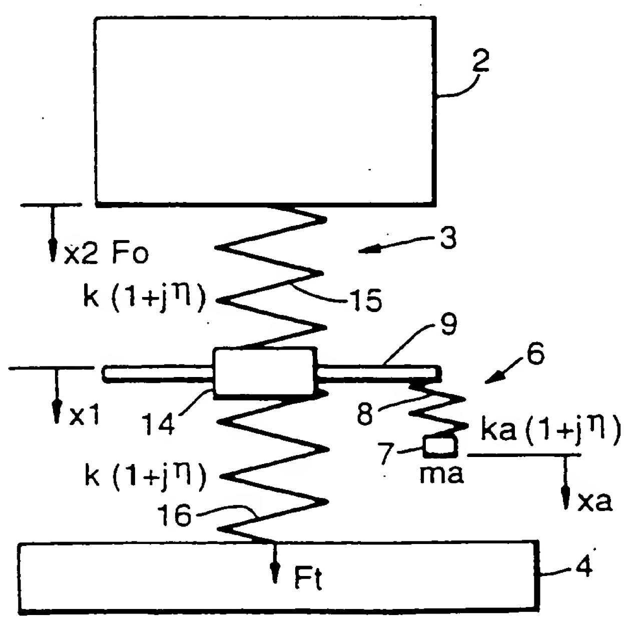

2
x2 Fo
3
k (1+jη)
15
9
6
x1
14
8
ka (1+jη)
7
ma
k (1+jη)
xa
16
Ft
4

Fig.9A.

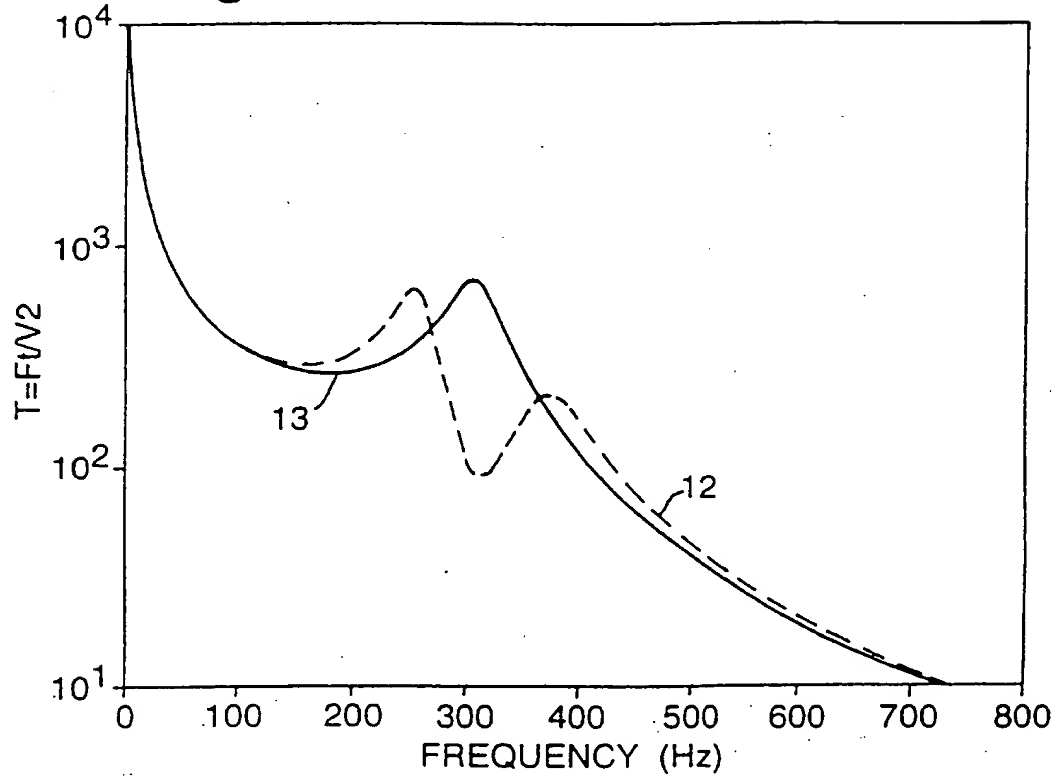

10⁴
10³
10²
10¹
T=Ft/V2
13
12
0
100
200
300
400
500
600
700
800
FREQUENCY (Hz)

Fig.9B.

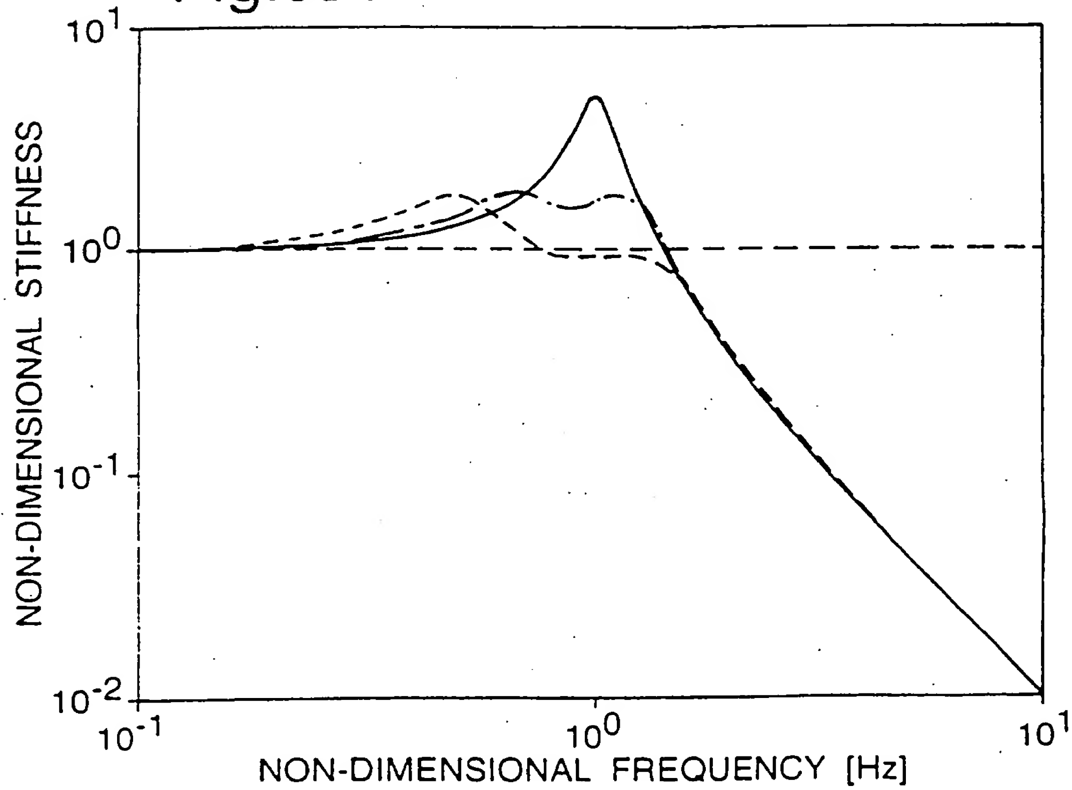

10¹
10⁰
10⁻¹
10⁻²
NON-DIMENSIONAL STIFFNESS
10⁻¹
10⁰
10¹
NON-DIMENSIONAL FREQUENCY [Hz]

Fig.10.

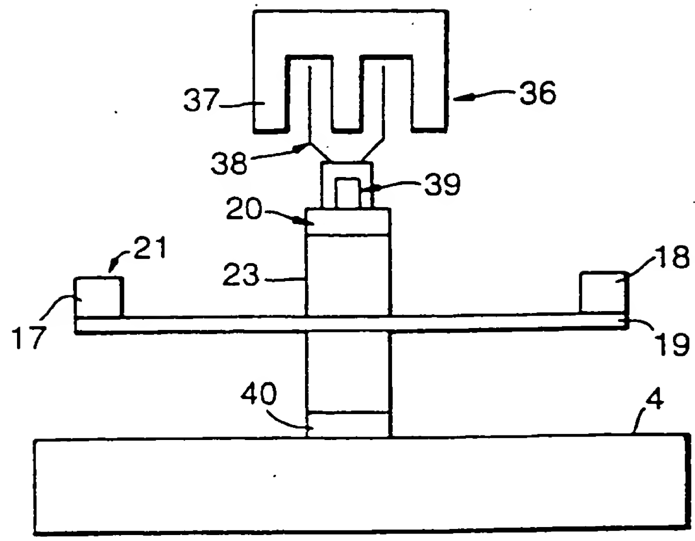


Fig.11.

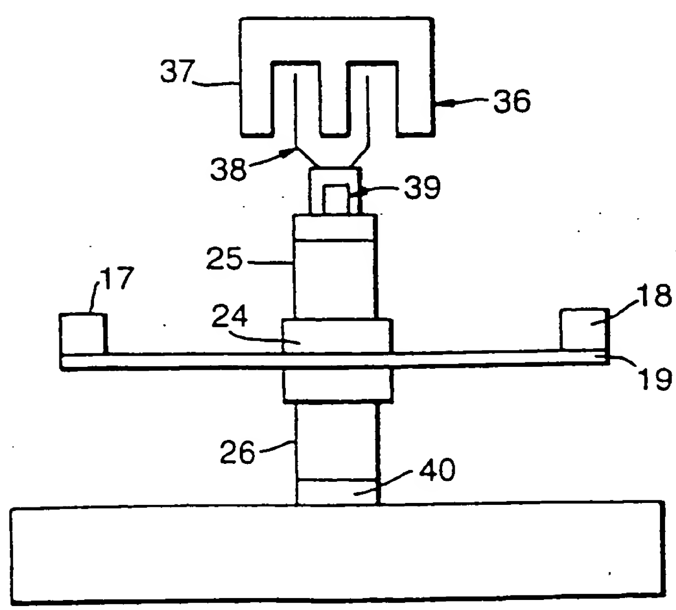


Fig.12.

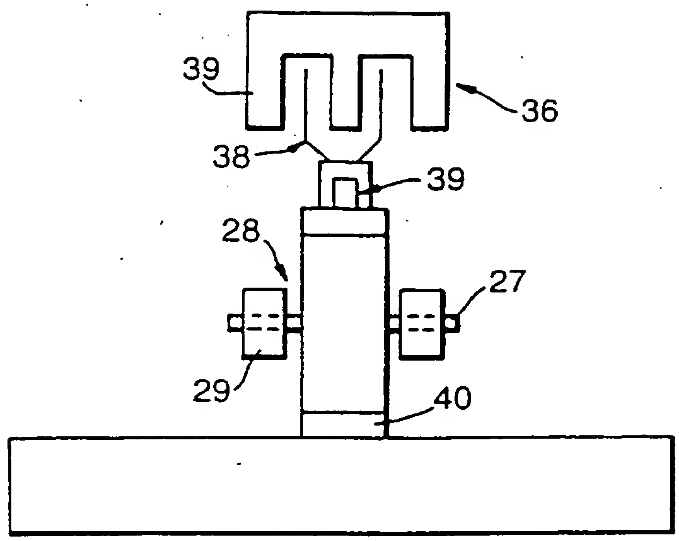


Fig.13.

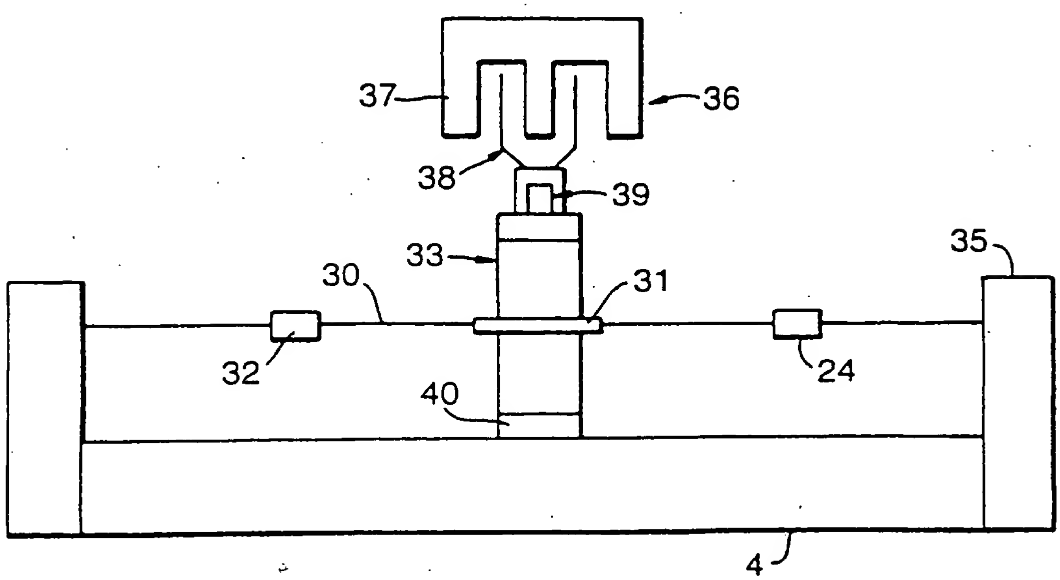
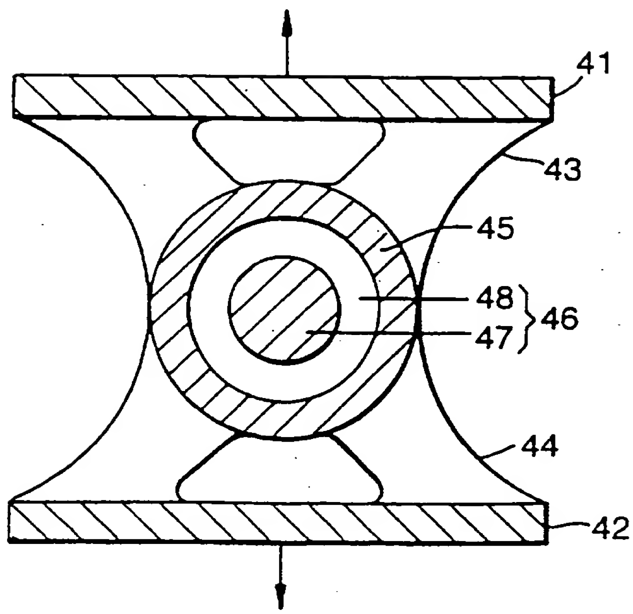


Fig.14.



INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 96/02332

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 F16F7/108 F16F1/371 F16F7/116

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F16F B64D B60K

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	DE 28 07 160 A (CONTINENTAL GUMMI WERKE AG) 30 August 1979 see the whole document ---	1-12,16, 17
X	PATENT ABSTRACTS OF JAPAN vol. 010, no. 291 (M-522), 3 October 1986 & JP 61 105320 A (MITSUBISHI HEAVY IND LTD), 23 May 1986, see abstract ---	1-6,10, 11,17
X	US 4 403 762 A (COGSWELL II JAMES A ET AL) 13 September 1983 see the whole document ---	1-6, 13-17
X	US 4 420 134 A (FLANNELLY WILLIAM G) 13 December 1983 see the whole document	1,2,5,6, 15,17
A		3
	--- -/--	

[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier document but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.
"&" document member of the same patent family

1

Date of the actual completion of the international search	Date of mailing of the international search report
20 January 1997	24.01.97

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax (+31-70) 340-3016	Van der Veen, F

Form PCT/ISA/210 (second sheet) (July 1992)

International Application No
PCT/GB 96/02332

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	SU 1 087 718 A (INST MASH IM A A BLAGONRAVOVA) 23 April 1984 see abstract; figure 1 ---	1-3,5,6, 15,17
A	EP 0 428 949 A (PIRELLI SISTEMI ANTIVIBRANTI) 29 May 1991 see the whole document -----	1,4-9, 13-16

1

INTERNATIONAL SEARCH REPORT

Information on patent family members

International Application No
PCT/GB 96/02332

Patent document cited in search report	Publication date	Patent family member(s)		Publication date
DE-A-2807160	30-08-79	NONE		
US-A-4403762	13-09-83	CA-A-	1172275	07-08-84
US-A-4420134	13-12-83	NONE		
SU-A-1087718	23-04-84	NONE		
EP-A-0428949	29-05-91	IT-B-	1237779	17-06-93
		DE-T-	69002153	17-02-94
		ES-T-	2047231	16-02-94
		JP-A-	3223541	02-10-91
		US-A-	5156380	20-10-92